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DEVELOPMENT OF AN EMULATION-SIMULATION THERMAL CONTROL MODEL FOR SPACE STATION APPLICATION

By

James G. Hartley
Gene T. Colwell

Submitted to:

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LANGLEY RESEARCH CENTER
HAMPTON, VIRGINIA 23665

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NASA Technical Officer
John B. Hall, Jr.
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GEORGIA INSTITUTE OF TECHNOLOGY
A UNIT OF THE UNIVERSITY SYSTEM OF GEORGIA
SCHOOL OF MECHANICAL ENGINEERING
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DEVELOPMENT OF AN EMULATION-SIMULATION THERMAL CONTROL MODEL FOR SPACE STATION APPLICATION

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ABSTRACT

The goal of this program is to develop an improved capability for comparing various techniques of thermal management in the "Space Station". The work involves three major tasks:

TASK I Develop a Technology Options Data Base.

TASK II Complete development of a Space Station Thermal Control Technology Assessment program.

TASK II Develop and evaluate emulation models.

INTRODUCTION

Current planning for the orbiting space station calls for a dual-keel configuration as shown in Figure 1. The thermal control system (TCS) for the space station is composed of a central TCS and internal thermal control systems for the modules, shown in Figure 2, as well as service facilities and attached payloads. The internal TCS may be attached to the central TCS through a thermal bus.

The central TCS is composed of a main transport system which collects waste thermal energy from each of the modules and transports it through coolant lines to the main rejection system. The main rejection system, in turn, is composed of steerable, constructable radiator elements attached to the transverse booms of the space station structure.

The waste heat loads arise from electrical and electronic equipment as well as metabolic loads in the manned modules. These equipment and metabolic loads may be collected by the central TCS, or they may be transported to small radiators mounted on the body of individual modules.

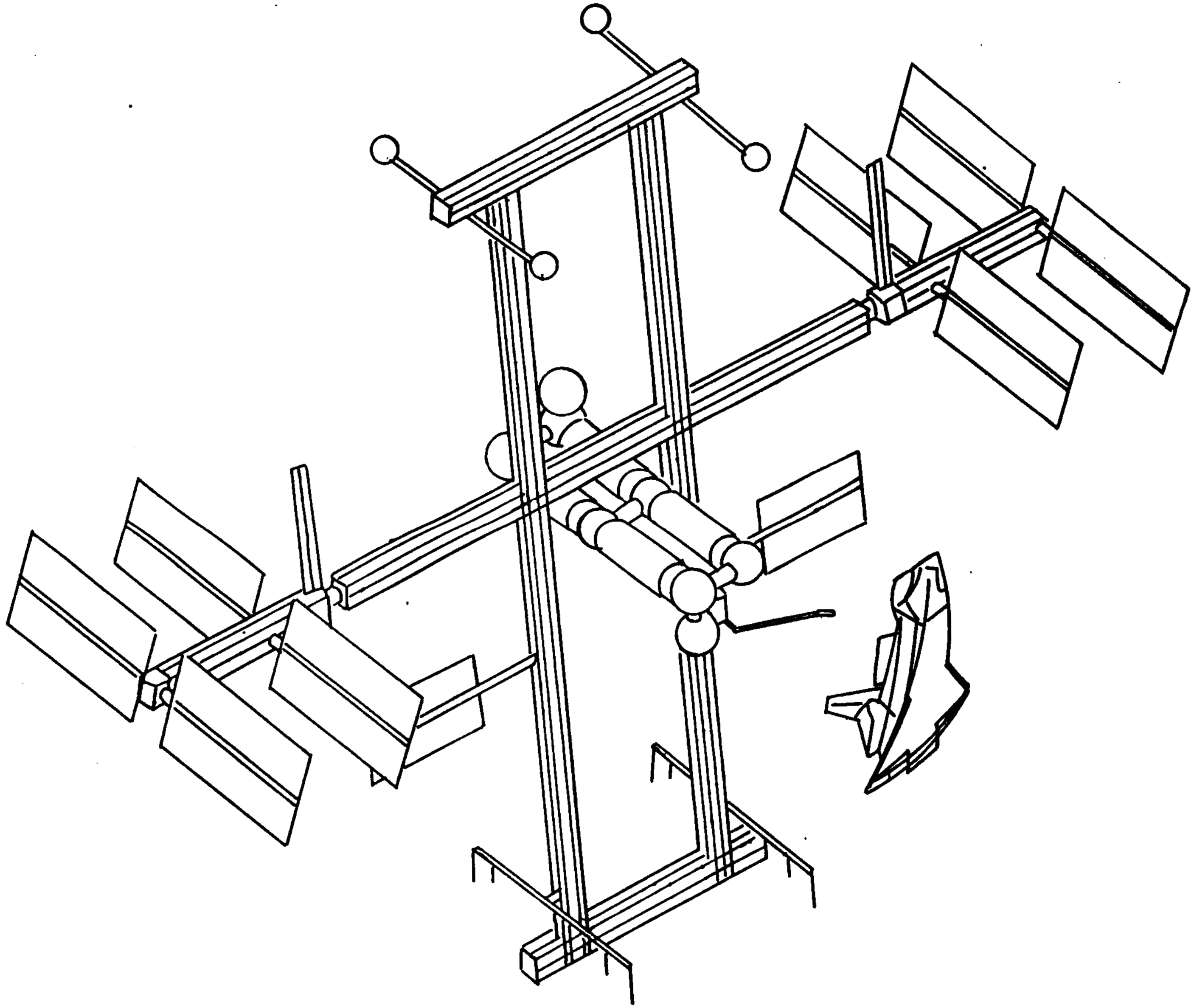

Figure 1. Space Station Configuration.

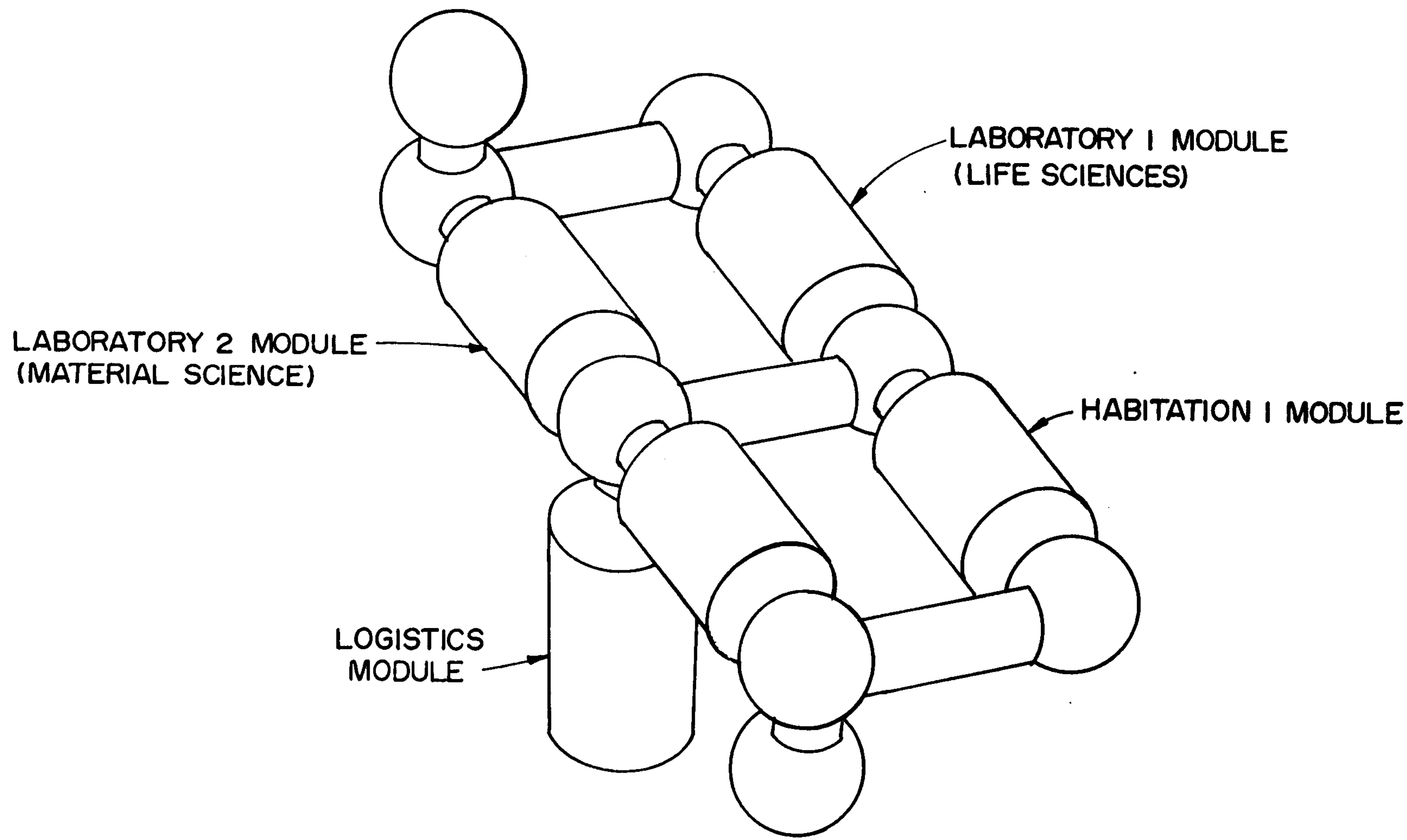

Figure 2. Station Modules.

Several candidate technologies are being considered for acquiring the waste heat loads, for transporting the thermal energy between the acquisition and rejection systems, and for rejecting the waste heat to space. The analysis techniques described here were developed for use in evaluating reliability, weights, costs, volumes, and power requirements for configurations using different candidates and different mission parameters.

EVALUATION TECHNIQUES

The thermal control system analysis program permits the user to design and analyze a space station thermal control system. The space station is assumed to be composed of seven distinct modules, and each may have its own metabolic heat loads and equipment heat loads. For each module, the user may specify the total metabolic load and the size and location of the equipment loads. The metabolic loads are assumed to be acquired by air-water heat exchangers, transported by pumped liquid water loops, and rejected to space by body-mounted radiators attached to each of the modules which have metabolic loads. Because the metabolic loop is local to a module it is called an autonomous loop.

Heat loads generated by equipment in each module are assumed to be acquired by cold plates. The user may choose among the following candidate technologies for the cold plates in each module:

1. Conductive cold plate
2. Two-phase cold plate
3. Capillary cold plate

In addition, the user may locate up to five cold plates (each having a different capacity) in a module, choose the cold plate operating temperature, and specify the working fluid (water, ammonia or Freon-11). The user also has the option to specify whether the equipment loop is to be integrated or autonomous. If the equipment loop is integrated, the heat from the equipment is transported from the cold plates to the main heat transport system for eventual rejection to space by the main rejection system. If the equipment loop is autonomous, the heat from the equipment is rejected to space by body-mounted radiators located on the module exterior. In this case the user may specify separate candidate technologies for heat transport and heat rejection in the autonomous equipment loop.

The user may select from the following candidate technologies for the main heat transport system or the heat transport system for a module having an autonomous equipment loop:

1. Pumped liquid loop
2. Pumped two-phase loop
3. High capacity heat pipe

In addition, the user may choose the transport lengths and specify the working fluid.

For the main heat rejection system or the heat rejection system for a module having an autonomous equipment loop, the user may select from the following candidate technologies:

1. Heat pipe radiator
2. High capacity heat pipe radiator
3. Liquid droplet radiator

In addition, the user may choose the radiator surface temperature and the emissivity of the radiator surface.

The data base for the thermal control system analysis program is divided into three major parts: the mission model parameters file, the candidate data files, and the system configuration file. Each is discussed in the following paragraphs. A detailed description of the data base contents is contained in Appendix A.

The mission model parameters file contains information which applies specifically to the mission or which applies to the space station as a whole. A sample mission model parameter file, as it appears to the user, is shown in Figure 3. When the program begins execution, the mission model parameter file is read from the data base. Any one or all of these parameters may be changed and used temporarily for assessment purposes or be replaced in the data base. In the latter instance, they become the new mission model parameter file when program execution begins anew because only the most recently saved version of the mission model parameter file is retained in the data base.

The candidate data files contain generic information for each of the candidate technologies available for heat acquisition, heat transport, and heat rejection. The data base contains one file for each candidate. A sample candidate data file, as it appears to the user, is shown in Figure 4. The weights, volumes, times, and costs shown in the figure are those for the specified candidate rating. If the candidate technology is used with a different rating, these values are scaled accordingly. When the program begins execution, the candidate data files are read from the data base. Any one or all of the values in these files may be changed and used

MISSION MODEL PARAMETERS

1.	M...MISSION DURATION, DAYS:	3650.00
2.	R...RESUPPLY INTERVAL,DAYS:	90.00
3.	NP..POWER PENALTY, LB/KW:	350.00
4.	NC..CONTROL PENALTY:	.00
5.	NP1.PROPULSION PENALTY:	60.00
6.	P...PROBABILITY OF METEROID PENETRATION, (0.920 TO 0.993):	.990
7.	CFA.TRANSPORTATION COST FACTOR, THOUSAND DOLLARS/LB:	1.60
8.	MR..MAINTENANCE COST FACTOR, THOUSAND DOLLARS/HR:	35.00
9.	IF..INTEGRATION COST FACTOR, %:	35.00
10.	PF..PROGRAMMATIC COST FACTOR, %:	70.00

DO YOU WISH TO CHANGE ANY VALUES (Y OR N)
DO YOU WISH TO REPLACE THE
MISSION MODEL PARAMETERS (Y OR N)

Figure 3. Mission Parameters.

LOGISTICS MODULE

ACQUISITIO SUBSYSTEM: CONDUCTIVE COLD PLATE

	TOTAL COLD PLATE CAPACITY, KW:			12.00
1.	NUMBER OF COLD PLATES:			3.00
2.	COLD PLATE OPERATING TEMPERATURE, C:			20.00
3.	METABOLIC LOAD, KW:			2.36
		CP #1	CP #2	CP #3
4.	HEAT REJECTION LOADS, KW:	4.00	4.00	4.00
5.	MAIN SUPPLY LINE LENGTHS, FT:	8.00	4.00	4.00
6.	BRANCH SUPPLY LINE LENGTHS, FT:	10.00	10.00	10.00
7.	MAIN RETURN LINE LENGTHS, FT:	8.00	4.00	4.00
8.	BRANCH RETURN LINE LENGTHS, FT:	10.00	10.00	10.00
9.	WORKING FLUID:			AMMONIA
	PIPE MATERIAL:			STAINLESS STEEL

DO YOU WISH TO CHANGE ANY VALUES (Y OR N)

Figure 5. Module Configuration Data.

8, 9

1. Acquisition assessment for each module
2. Summary acquisition assessment for all modules
3. Summary transport assessment for the main transport system
4. Summary rejection assessment for the main rejection system
5. Summary assessment for the entire thermal control system.

The analysis begins with a determination of the launch weight, launch volume, heat transfer surface areas, and external power requirement imposed by the acquisition system for each module. These computations depend upon the acquisition candidate and module configuration and are performed in separate subroutines - one for each of the candidate technologies. For example, acquisition system subroutines contain algorithms for sizing coolant lines for minimum weight, determining cold plate sizes and weights, computing pumping power required, determining thermal bus connection requirements, and computing the volume occupied by the acquisition systems. These computations depend upon the candidate technology employed (i.e. single-phase or two-phase cold plates, etc.), working fluid, materials, and operating temperatures. For a rejection system candidate such as a heat pipe radiator, the candidate subroutine contains algorithms for assessing the performance of heat pipe elements which would be used to construct the radiator. In this case, parameters such as working fluid, material, radiator temperature, geometry, and surface radiative properties may be selected and included in the design calculations.

The launch weight, launch volume, surface areas, and power requirement computed in the candidate subroutine, together with the mission model parameters and candidate data file, are used to compute all of the other assessment information. The algorithms for these computations are detailed in Appendix B. A flow schematic illustrating the operation of the program as the user views it is shown in Figure 7. The following paragraphs describe several of the thermal models used in the candidate subroutines.

CONDUCTIVE COLD PLATE MODEL (Subroutine CCP)

The conductive cold plate is assumed to have an equipment mounting face of length L and width W. The cold plate has n channels for liquid flow, each of which has a hydraulic diameter of D_H. The power, Q, dissipated by the equipment mounted on the cold plate is assumed to be uniformly distributed over the surface of the cold plate. The cooling fluid enters the cold plate at temperature T_i and leaves at temperature T_o. The cold plate operating temperature is T_p, and T_f is the average temperature of the fluid in the cold plate. The temperature difference (T_p-T_f) is assumed to be the same for all operating conditions.

The total mass flow rate, $\dot{m}$, of fluid in the cold plate is computed from the following expression:

$$\dot{m} = \frac{Q}{c_p(T_o - T_i)} \qquad (1)$$

The temperature difference (T_o-T_i) is assumed to be the same for all operating conditions.

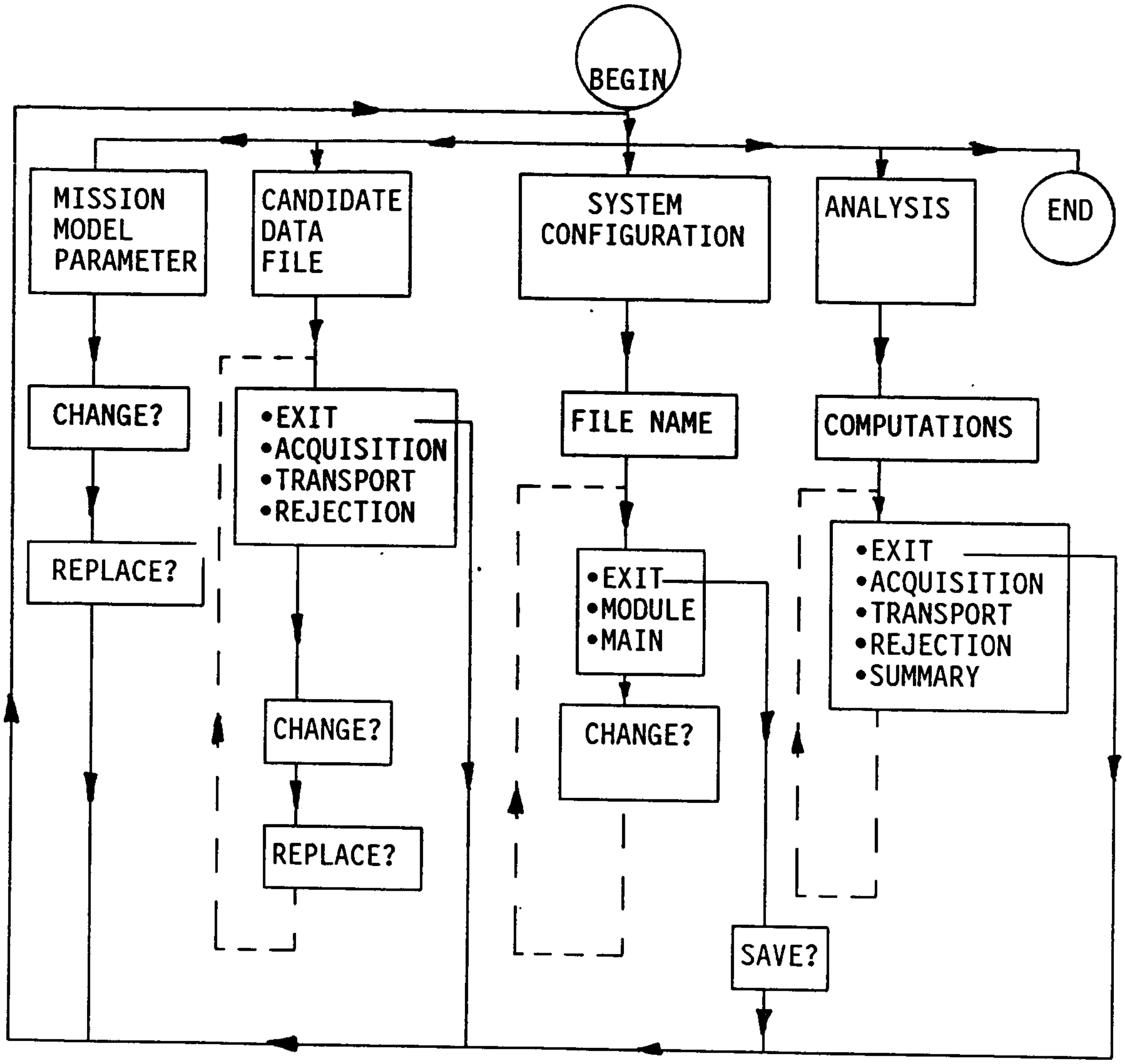

Figure 7. TCS PROGRAM SCHEMATIC

For a specific cold plate design, the ratio of the plate surface area to the internal wetted perimeter is assumed to be constant, i.e.

$$\frac{A_o}{n\pi D_H L} = \text{constant} \tag{2}$$

and the hydraulic diameter and length of each flow passage are assumed to be fixed. The fluid flow through the internal channels is assumed to be turbulent, and the inside convective heat transfer coefficient is determined by [1]

$$h = \frac{0.023\ f(T)\ V^{0.8}}{D_H^{0.2}} \tag{3}$$

where f(T) accounts for the temperature dependence of the fluid properties:

$$f(T) = \frac{k^{0.67}(\rho c)^{0.33}}{\nu^{0.47}}$$

Furthermore, the mass flow rate is related to the fluid velocity through the continuity equation:

$$\dot{m} = \frac{\rho n \pi D_H^2 V}{4} \tag{4}$$

where n is the number of parallel passages, or internal channels, in the cold plate. The heat flux at the cold plate surface is computed from

$$q'' = \frac{Q}{A_o} \tag{5}$$

where A_o is the area of the mounting surface. The heat flux is also related to the difference between the cold plate surface temperature and the average fluid temperature by the expression

$$q'' = \frac{U_i n \pi D_H L(T_p - T_f)}{A_o} \qquad (6)$$

where U_i is the overall heat transfer coefficient based on the inside surface area of a single flow passage. This coefficient is computed as

$$U_i = \left[\frac{1}{h} + \frac{\delta}{k_m}\right]^{-1}$$

where δ is a characteristic path length for conduction through the cold plate material from the interior wall of the flow passage to the cold plate external surface. Equations (1) through (6) can be written in the following dimensionless forms with the aid of reference values, denoted by the superscript *, which are determined from a specific set of design conditions:

$$\frac{\dot{m}}{\dot{m}^*} = \frac{Q\, c_p^*}{Q^* c_p} \qquad (8)$$

$$\frac{A_o}{A_o^*} = \frac{n}{n^*} \qquad (9)$$

$$\frac{h}{h^*} = \frac{f(T)}{f(T^*)} \left[\frac{V}{V^*} \right]^{0.8} \tag{10}$$

$$\frac{\dot{m}}{\dot{m}^*} = \frac{\rho Vn}{\rho^* V^* n^*} \tag{11}$$

$$\frac{q''}{q''^*} = \frac{QA_o^*}{Q^* A_o} \tag{12}$$

$$\frac{q''}{q''^*} = \frac{U_i}{U_i^*} \tag{13}$$

In these equations, parameters without a superscript are those for the new set of operating conditions. Next, equations (8) through (13) can be combined to produce the following transcendental equation for the velocity of the fluid through each flow passage.

$$V = \frac{\rho^* c_p^* V^*}{\rho c_p U_i \left[\frac{f(T^*)}{h^* f(T)} \left[\frac{V^*}{V} \right]^{0.8} + \frac{\delta}{k_m} \right]} \tag{14}$$

With the fluid velocity known, the overall heat transfer coefficient can be computed from

$$U_i = U_i^* \frac{\rho c_p V}{\rho^* c_p^* V^*}$$

This expression is obtained by combining Eqs.(8), (9), and (11) through (13). Next, the surface heat flux can be determined from Eq. (13), and the heat transfer surface area required for the new operating conditions can be computed from Eq. (5). Because the ratio of the plate surface area to the internal wetted perimeter is assumed constant, the ratio of the cold plate volume to the plate surface area is also assumed constant,

$$\frac{VOL}{A_o} = \text{constant} = c_1 \tag{15}$$

Thus, the volume can be determined once the surface area is known. In addition, the weight of the cold plate is directly proportional to the cold plate volume and the density of the cold plate material

$$W = c_2 \rho_m \, VOL = c_1 c_2 \rho_m A_o \tag{16}$$

By combining Eqs. (15) and (16), we obtain an expression for the weight of the cold plate in terms of surface area,

$$W = A_o \left(\frac{W^*}{A_o^*} \right) \left(\frac{\rho_m}{\rho_m^*} \right) \tag{17}$$

The analysis presented here is incorporated in subroutine CCP, and the reference values for this analysis are listed in Table 1.

TABLE 1. Reference Design Values for Conductive Cold Plate Analysis.

Variable	Value	Reference
Q^*	10 kW	
q''^*	0.27 kW/ft^2	2
$\dot{m}^*$	1.0542 lb/s	
U_i^*	298.7 $Btu/hr\text{-}ft^2\text{-}^oF$	
V^*	0.387 m/s	
T^*	20^oC	2
h^*	364 $Btu/hr\text{-}ft^2\text{-}^oF$	
(T_o-T_i)	5^oC	2
δ	0.005 ft	
C_1	0.0292 ft	
W^*/A^*	5.3 lb/ft^2	2
ρ_m^*	488 lb/ft^3 (Type 304 SS)	1
k_m^*	8.319 $Btu/hr\text{-}ft\text{-}^oF$ (Type 304 SS)	1
$\rho^*, c_p^*, \nu^*, k^*$	evaluated for water at 20^oC	

TWO-PHASE COLD PLATE MODEL (Subroutine TPCP)

The two-phase cold plate is assumed to have an equipment mounting face of length L and width W. The cold plate has n channels for fluid flow, each of which has a hydraulic diameter of D_H. The power, Q, dissipated by the equipment mounted on the cold plate is assumed to be uniformly distributed over the surface of the cold plate. The cooling fluid enters the cold plate as a saturated liquid at temperature T_f and leaves at temperature T_f with a quality of $\dot{X}$. The cold plate operating temperature is T_p, and the temperature difference $(T_p - T_f)$ is assumed to be the same for all operating conditions. The total mass flow rate, $\dot{m}$, of fluid in the cold plate is computed from the following expression:

$$\dot{m} = \frac{Q}{X\ h_{fg}} \tag{1}$$

The quality at the exit is assumed to be the same for all operating conditions. For a specific cold plate design, the ratio of the plate surface area to the internal wetted perimeter is assumed to be constant, i.e.

$$\frac{A_o}{n\pi D_H L} = \text{constant} \tag{2}$$

and the hydraulic diameter and length of each flow passage are assumed to be fixed. The inside convective heat transfer coefficient is determined by [3]

$$h = 9.0 \times 10^{-4} f(T) G \tag{3}$$

where the mass flux, G, is determined from

$$G = \frac{4\,\dot{m}}{n\pi D_H^{\,2}} \tag{4}$$

n is the number of parallel passages, or internal channels, in the cold plate, and f(T) accounts for the temperature dependence of the fluid properties:

$$f(T) = \frac{k_l\ k_f^{\,1/2}}{\mu_l}$$

where K_f is the boiling number defined as

$$K_f = \frac{X\ h_{fg}}{gL}$$

The heat flux at the cold plate surface is computed from

$$q'' = \frac{Q}{A_o} \tag{5}$$

where A_o is the area of the mounting surface. The heat flux is also related to the difference between the plate surface temperature and the average fluid temperature by the expression

$$q'' = \frac{U_i n\pi D_H L(T_p - T_f)}{A_o} \tag{6}$$

where U_i is the overall heat transfer coefficient based on the inside surface area of a single flow passage. This coefficient is computed as

$$U_i = \left[\frac{1}{h} + \frac{\delta}{k_m} \right]^{-1} \tag{7}$$

where δ is a characteristic path length for conduction through the cold plate material from the interior wall of the flow passage to the cold plate external surface. Equations (1) through (6) can be written in the following dimensionless forms with the aid of reference values, denoted by the superscript *, which are determined from a specific set of design conditions:

$$\frac{\dot{m}}{\dot{m}^*} = \frac{Q\, h_{fg}^*}{Q^*\, h_{fg}} \tag{8}$$

$$\frac{A_o}{A_o^*} = \frac{n}{n^*} \tag{9}$$

$$\frac{h}{h^*} = \frac{f(T)\, G}{f(T^*) G^*} \tag{10}$$

$$\frac{G}{G^*} = \frac{\dot{m} n^*}{\dot{m}^* n} \tag{11}$$

$$\frac{q''}{q''^*} = \frac{Q\, A_o^*}{Q^* A_o} \tag{12}$$

$$\frac{q''}{q''^*} = \frac{U_i}{U_i^*} \tag{13}$$

In these equations, parameters without a superscript are those for the new set of operating conditions. Next, equations (8) through (13) can be combined to produce the following equation for the mass flux of the fluid through each flow passage

$$G = \frac{k_m}{\delta} \left[\frac{G^* h_{fg}^*}{U_i^* h_{fg}} - \frac{f(T^*)G^*}{f(T)h^*} \right] \tag{14}$$

With the mass flux known, the overall heat transfer coefficient can be computed from

$$U_i = U_i^* \frac{G h_{fg}}{G^* h_{fg}^*}$$

This expression is obtained by combining Eqs.(8), (9), and (11) through (13). Next the surface heat flux can be determined from Eq. (13), and the heat transfer surface area required for the new operating conditions can be computed from Eq. (5). Because the ratio of the plate surface area to the internal wetted perimeter is assumed constant, the ratio of the cold plate volume to the plate surface area is also assumed constant,

$$\frac{VOL}{A_0} = C_1 \tag{15}$$

Thus, the volume can be determined once the surface area is known. In addition, the weight of the cold plate is directly proportional to the cold plate volume and the density of the cold plate material

$$W = C_2 \rho_m VOL \tag{16}$$

The analysis presented here is incorporated in subroutine TPCP, and the reference values for this analysis are listed in Table 2.

HIGH CAPACITY HEAT PIPE RADIATOR MODEL (Subroutine CANDR2)

A high performance heat pipe radiator using a series of heat pipes with combination slab and circumferential capillary structure is modeled for space station use in the temperature range of 310 K to 366 K (100°F to 200°F). A schematic of the capillary structure is shown in Figure 8. Axial transport of working fluid primarily occurs through the central slab while the circumferential structure distributed the fluid around the circumference in the heated and cooled sections.

TABLE 2. Reference Design Values for Two-Phase Cold Plate Analysis.

Variable	Value	Reference
Q^*	5 kW	
q''^*	0.6 kW/ft^2	4
$\dot{m}^*$	17.97 lb/hr	
U_i^*	296.4 Btu/hr-ft^2-^{o}F	
G^*	1.5 x 10^4 lb/ft^2-hr	
T^*	20^{o}C	4
h^*	377 Btu/hr-ft^2-^{o}F	
δ	0.006 ft	
C_1	0.0833 ft	
C_2	0.22	
ρ_m^*	488 lb/ft^3 (Type 304 SS)	1
k_m^*	8.319 Btu/hr-ft-^{o}F (Type 304 SS)	1
$\rho^*, h_{fg}^*, \mu^*, k^*$	evaluated for water at 20^{o}C	

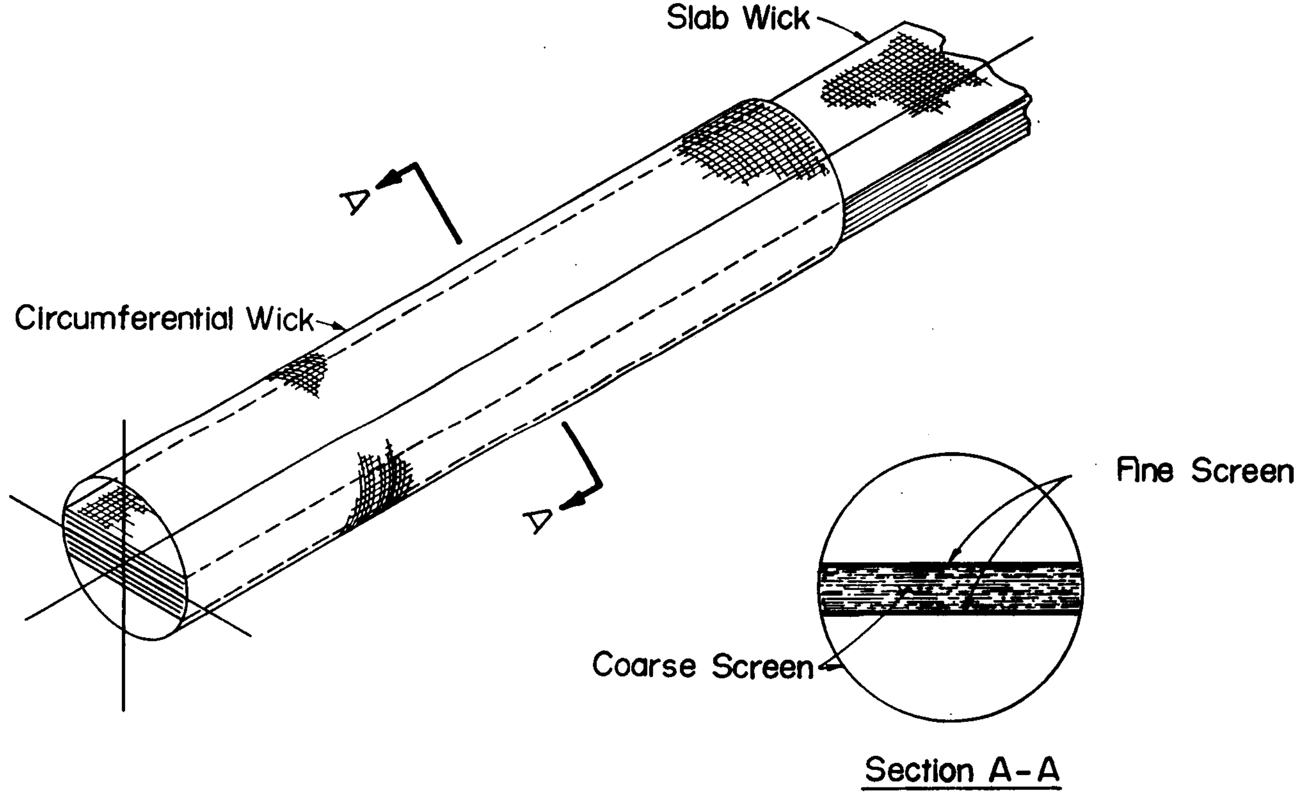

Figure 8. Composite Slab and Circumferential Capillary Structure at Evaporator.

Performances of various heat pipes to be used in a radiator panel are estimated from experimental studies performed at Georgia Tech, on a Refrigerant-11 heat pipe with slab capillary structure [5]. This heat pipe can transport a maximum of about 130 watts of thermal energy at 440 K when operating with Refrigerant-11 as the working fluid. Heat pipes to be used in radiators for the space station, may use other working fluids, may utilize different capillary structures, may be of different outside diameter and/or length, and may operate at different temperatures. All of these design parameters greatly affect heat pipe thermal transport capacity.

Writing momentum, energy, and continuity equations for steady operation of the model heat pipe at capillary limited heat transfer and making the standard simplifying assumptions, the following equation is obtained [6].

$$\dot{Q}_{CL} = \frac{2N/r_p}{\frac{\bar{K}L_{eff}}{b\delta_T} + \frac{K_c L}{4n_c\delta_c}\left(\frac{1}{L_e} + \frac{1}{L_c}\right) + \frac{8\mu_V\rho_L L_{eff}}{\pi\mu_L\rho_V r_V^4}}$$

where

$\dot{Q}_{CL}$ = Capillary limited heat transfer rate

$N = \frac{\sigma h_{fg}\rho_L}{\mu_L}$ = "Heat Pipe Number"

σ = surface tension of liquid

h_{fg} = heat of vaporization

ρ_L, ρ_V = liquid density, vapor density

μ_L, μ_V = liquid dynamic viscosity, vapor dynamic viscosity

r_p = pore radius at evaporator surface

$\bar{K} = \dfrac{\delta_T}{\dfrac{n_A \delta_A}{K_A} + \dfrac{n_B \delta_B}{K_B}}$ = effective inverse permeability for slab based on approach velocity.

δ_T = total thickness of slab

n_A = number of layers of fine mesh in slab

n_B = number of layers of coarse mesh in slab

δ_A = thickness of a single layer of material A

δ_B = thickness of a single layer of material B

K_A = inverse permeability for material A based on approach velocity

K_B = inverse permeability for material B based on approach velocity

L_{eff} = effective length of liquid path in slab

b = width of slab

K_C = inverse permeability for material at evaporator and condenser surfaces based on approach velocity

L = average distance traveled by liquid in circumferential capillary structure at evaporator or condenser (approximately 45^o arc)

n_C = number of layers of capillary material on circumference

δ_C = thickness of a single layer of material C

L_e = axial length of evaporator section

L_c = axial length of condenser section

r_V = hydraulic radius of vapor space

The three terms in the denominator of this equation are related to flow resistance in the central slab, the circumferential capillary structure, and the vapor region, respectively. For the present design, flow resistance is much larger in the slab than in the circumferential structure or in the vapor region. Thus, approximately

$$\dot{Q}_{CL} \simeq \frac{2N}{\frac{r_p \bar{K} L_{eff}}{b\delta_T}}$$

and

$$\dot{Q}_{CL_{II}} = \dot{Q}_{CL_I} \frac{N_{II}}{N_I} \frac{\bar{K}_I}{\bar{K}_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{L_{eff,I}}{L_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

where subscript I refers to a known performance and known design parameters and II refers to predicted performance when new design parameters are chosen. The width of the slab is assumed constant.

Design heat transport capability is assumed to be one-half of maximum transport capability.

$$\dot{Q}_D = \dot{Q}_{CL}/2$$

and therefore the design heat transport is given by

$$\dot{Q}_{D_{II}} = \dot{Q}_{D_I} \frac{N_{II}}{N_I} \frac{\bar{K}_I}{\bar{K}_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{L_{eff,I}}{L_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

The following design parameters for the radiator are chosen:

Heat load 50, kW

Steerable radiator with thermal storage

Absorptivity, $\alpha_s = 0.30$

Emissivity, $\epsilon = 0.78$

Heat pipe fluid at $100^{o}F$

Radiator average surface temperature, $75^{o}F$

Area, 2,500 ft^2

Material, aluminum

Figure 9 shows a radiator constructed from a series of 50 foot heat pipe and fin panels. Assuming each heat pipe is 3/4-in. outside diameter, 5/8-in. inside diameter, and 50 feet long, the metal weight will be about 8 lbm and the working fluid will weigh about 1.5 lbm for a total weight of 9.5 lbm per pipe. The panel width and weight per panel are given by the following expressions:

$$w_p \text{ (in)} = \text{panel width} = \frac{631}{N_p}$$

$$m_p\text{(lbm)} = \text{weight per panel}$$

$$= 600/N_p\ [631 - N_p(0.75)](0.0625)(0.1) + 9.5$$

where N_p is the number of heat pipes in 50 kW radiator and the fin thickness is taken to be 1/16 inch.

Table 3 shows the results of selecting different working fluids and working fluid temperatures. The parameters used in

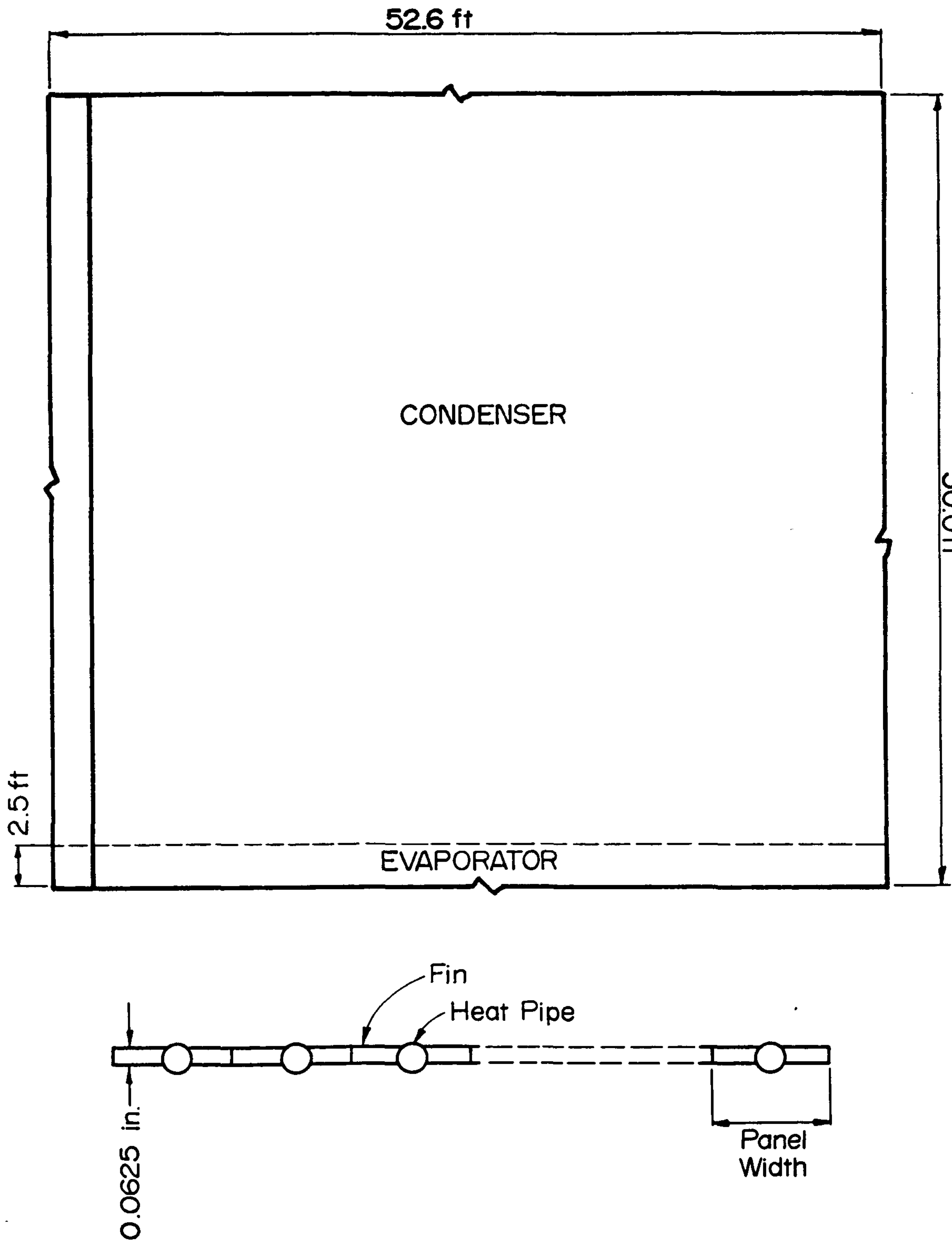

Figure 9. Heat Pipe Radiator.

TABLE 3. Heat Pipe Radiator Design Results

Parameter	Heat Pipe Working Fluid and Temperature R-11 310 K	R-11 366 K	Methanol 310 K	Methanol 366 K	Ammonia 310 K	Ammonia 366 K	Acetone 310 K	Acetone 366 K
$\dot{Q}_{CL}$(kW)	0.440	0.367	1.54	1.61	2.03	0.660	1.10	0.918
$\dot{Q}_{D}$(kW)	0.220	0.184	0.770	0.805	1.015	0.330	0.550	0.459
Number of Pipes for 50 kW	229	275	65	62	49	153	92	110
Panel Width Per Pipe (in)	2.62	2.18	9.23	9.68	12.24	3.92	6.52	5.45
Weight Per Panel (lbm)	16.5	14.9	41.3	43.0	52.6	21.4	31.1	27.1
Total Radiator Weight (lbm)	3,780	4,090	2,690	2,660	2,580	3,270	2,870	2,990
Radiator Volume (ft^3)	156	156	156	156	156	156	156	156

computing values listed in the table are shown in Table 4. Design heat transfer per pipe (taken to be one half of capillary limitation) ranges between about 1 kW for ammonia at 310 K to about 0.18 kW for R-11 at 366 K, while total radiator weight varies between 2,580 lbm for ammonia at 310 K to 4,090 lbm for R-11 at 366 K.

The following equations may be used to predict areas and weights for a particular candidate from known values for the base design.

A. Design Heat Transport Per Pipe

$$\dot{Q}_{D_{II}} = \dot{Q}_{D_I} \frac{N_{II}}{N_I} \frac{\bar{K}_I}{\bar{K}_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{L_{eff,I}}{L_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

where subscripts I and II refer to the base case and case to be computed, respectively.

B. Number of Panels

$$N_P = \frac{\dot{Q}}{\dot{Q}_{D_{II}}}$$

where $\dot{Q}$ = radiator rating (kW)

C. Radiator Surface Area

$$\frac{A_{II}}{A_I} = \frac{\dot{Q}_{II}}{\dot{Q}_I} \frac{\epsilon_I}{\epsilon_{II}} \frac{F_{\alpha II}}{F_{\alpha I}} \left(\frac{T_I}{T_{II}} \right)^4$$

where

$F_\alpha = 1 + 0.5\ (\alpha_s - 0.20)$, adapted from reference [7] page 525

and

$F_{\alpha I} = 1 + 0.5\ (0.30 - 0.20) = 1.05$

TABLE 4. **Heat Pipe Base Design - Georgia Tech Heat Pipe.**

Parameters	Values
Rating	50 kW
Area	2500 ft^2 - reference [8]
Radiator surface temperature	297 K
Material	aluminum
Heat pipe I.D.	0.625 in.
Heat pipe O.D.	0.75 in.
Fin thickness	0.0625 in.
Heat pipe length	50 ft.
Evaporator length	2.5 ft.
Condenser length	47.5 ft.
Working fluid	ammonia
Working fluid temperature	310 K
Design heat transfer per pipe	1.02 kW
Number of panels	50
Panel width per pipe	12.24 in.
Capillary structure - 2 layers 400 mesh on circumference, 4 layers 400 mesh + 5 layers 30 mesh in slab	
Weight per panel	52.6 lbm
Total radiator weight (exclusive of heat exchanger)	2,580 lbm
Radiator volume (exclusive of heat exchanger)	156 ft^3
Absorptivity, α_S	0.30
Emissivity, ϵ	0.78
Ratio α_S/ϵ	0.385
$\dot{K}_I$, effective inverse permeability of slab	0.696×10^9 $(1/m^2)$
r_{p_I} pore radius at evaporator,	1.91×10^{-5} m
$L_{eff,I}$ heat pipe effective length,	25 ft
N_I, heat pipe number,	5.6×10^{10} W/m^2
δ_{T_I}, slab total thickness,	3.41×10^{-3} m

D. Radiator Width

Assuming a length of 50 ft. for each panel, the radiator total width is given by

$$W_R(ft) = \frac{A_{II}(ft)^2}{50}$$

E. Width Per Panel

$$W_p(ft) = \frac{W_R(ft)}{N_p}$$

F. Weight Per Panel

$$m_p(lbm) = 0.0217\ \rho_m[12\ W_R - N_p\ (0.75)]/N_p + 1.5 + \rho_m/21.8$$

G. Total Radiator Weight (excluding heat exchangers)

$$m_R(lbm) = m_p N_p$$

H. Total Radiator Volume

$$V_R(ft^3) = 0.26\ W_R$$

These equations have been incorporated into subroutine CANDR2 in the thermal control system analysis program.

SIZING LIQUID LINES (Subroutine LIQLINE)

The pipe sizes for liquid supply or liquid return lines are determined by minimizing the weight of the piping system [2]. Each segment of pipe in the longest pipe run is optimized individually by minimizing its mass which is determined from

$$\text{Mass} = M_i = \text{mass of pipe} + \text{mass of liquid} + \text{pump power penalty mass}$$

where

$$\text{mass of pipe} = \rho_{ss} L_i \pi (D_i + t_i) t_i$$

$$\text{mass of liquid} = \rho_L \pi D_i^2 L_i / 4$$

$$\text{pump power penalty mass} = M_p P_p$$

The pump power penalty is $\dot{M}_p$ (lb/kW), and the pump power is determined from

$$P_p = \frac{\dot{m}_i \Delta P_i}{\rho_L \eta_p}$$

The pressure drop for the segment of pipe is calculated from

$$\Delta P_i = \frac{8 L_i \dot{m}_i^2 f_i}{\pi^2 \rho_L D_i^5}$$

where the friction factor for turbulent flow in smooth pipes [8] is

$$f_i = 0.316/Re^{1/4}$$

and for laminar flow [10] is

$$f_i = 64/Re$$

The Reynolds number is defined as

$$Re = \frac{4\ \dot{m}_i}{\pi \mu_L D_i}$$

Thus

$$\Delta P_i = \frac{128\ \mu_L L_i \dot{m}_i}{\pi \rho_L D_i^4}$$

and the pipe segment mass to be minimized is

$$M_i = \rho_{ss} L_i \pi (D_i + t_i) t_i + \rho_L \pi D_i^2\ L_i/4 + M_p \frac{\dot{m}_i \Delta P_i}{\rho_L \eta_p}$$

The pipe thickness, t_i, is determined by the internal pipe diameter according to standard pipe and tube specifications.

SIZING VAPOR LINES (Subroutine VAPLINE)

The vapor line sizes in two-phase systems are selected consistent with the desire to limit the loss of stagnation pressure and stagnation temperature in vapor return lines [1]. The analysis of these losses is based upon adiabatic, compressible pipe flow with friction [11] as outlined below.

The vapor line diameter for each pipe segment in the vapor return line is chosen such that the stagnation pressure drop is less than 2 percent of the stagnation pressure at the exit of the cold plate. The conditions at the inlet of the vapor line are denoted by the subscript 1 and the subscript 2 denotes the conditions at the exit. We require that

$$P_{02}/P_{01} \geq 0.98 \qquad (6)$$

where the zero subscript designates stagnation conditions.

The stagnation pressure ratio can be computed from

$$\frac{P_{02}}{P_{01}} = \frac{M_1}{M_2} \left[\frac{(1 + \frac{k-1}{2} M_2{}^2)}{(1 + \frac{k-1}{2} M_1{}^2)} \right]^{\frac{(k+1)}{2(k-1)}}$$

where

$M_i = V_i/C_i$ is the Mach number

$C_i = \sqrt{kRT_i g_c}$ is the sonic velocity

$k = c_p/c_v$ is the ratio of specific heats for the vapor

R is the gas constant for the vapor

The general procedure for determining the information necessary to calculate the stagnation pressure ratio is iterative in nature as outline in the following.

1. Assume a pipe diameter D and calculate the inlet vapor velocity, V_1, from the know mass flow rate.
2. Calculate the inlet Mach number, M_1

3. Calculate the inlet Reynolds number, Re_1, determine the friction factor, f, for turbulent or laminar flow as dictated by the Reynolds number, and calculate $\bar{f}L/D)_{actual}$ from the given pipe length and assumed diameter.

4. Calculate the inlet stagnation temperature

$$T_{01} = T_1 + \frac{V_1^2}{2C_p}$$

and the inlet stagnation pressure

$$P_{01} = P_1 \left(\frac{T_{01}}{T_1} \right)^{k/(k-1)}$$

5. Calculate the quantity $\bar{f}L^*/D)_1$ at the inlet,

$$\left.\frac{\bar{f}L^*}{D}\right)_1 = \frac{1 - M_1^2}{k\, M_1^2} + \frac{k+1}{2k} \ln \left[\frac{(k+1)M_1^2}{2[1 + \frac{1}{2}\,(k-1)M_1^2]} \right]$$

and the quantity $\left.\frac{\bar{f}L^*}{D}\right)_2$ from

$$\left.\frac{\bar{f}L^*}{D}\right)_2 = \left.\frac{\bar{f}L^*}{D}\right)_1 - \left.\frac{\bar{f}L}{D}\right)_{actual}$$

6. Solve the following transcendental equation for the exit Mach number, M_2:

$$\left.\frac{\bar{f}L^*}{D}\right)_2 = \frac{1 - M_2^2}{kM_2^2} + \frac{k+1}{2k} \ln \left[\frac{(k+1)M_2^2}{2[1 + \frac{1}{2}\,(k-1)M_2^2]} \right]$$

7. Finally, compute P_{02}/P_{01} from Equation (6). If $P_{02}/P_{01} < 0.98$, choose a large pipe diameter and repeat steps 1 through 6. If $P_{02}/P_{01} > 0.98$ choose a smaller pipe diameter and repeat steps 1 through 6. If $P_{02}/P_{01} \simeq 0.98$, the assumed pipe diameter is adequate for this pipe segment.

EQUIPMENT LOOPS WITH CONDUCTIVE COLD PLATES (Subroutine CANDA1)

Equipment loops with conductive cold plates employ a working fluid that remains in the liquid phase. The analysis of these loops is performed in subroutine CANDA1 as outlined below.

1. The metabolic loop is analyzed using subroutine METLOOP to to determine the volume, mass, and pump power for the metabolic loops.
2. The conductive cold plates in the equipment loop are analyzed using subroutine CCP to determine the mass flow rates through each cold plate, the mass flow rates through each segment of the liquid supply and liquid return lines, the total acquisition surface area, the total cold plate mass, and the total cold plate volume.
3. The liquid supply lines, the liquid return lines, and the branch lines are sized using subroutine LIQLINE to determine the pipe mass, the fluid mass, the piping volume, and the total pressure drop in the equipment loop. (The pressure drop through each cold plate is assumed to be 5 psi.)

4. The total pump power requirement for the equipment loop is determined in subroutine DELPRS.

5. The weight of the pump package for the equipment loop and for the metabolic loop is computed.

6. The results of these analyses are stored in the TEMP array in the following order where IMOD denotes the module number or index:

   TEMP(IMOD,1) = pump power required, kW

   This value includes the pump power required for the equipment loop and the pump power required by the metabolic loop.

   TEMP(IMOD,2) = total mass, lb

   This value includes the cold plate mass, the dry pipe mass and the fluid mass of the equipment loop, the total mass (wet pipe and heat exchanger) of the metabolic loop, and the pump package weight for the equipment loop and for the metabolic loop.

   TEMP(IMOD,3) = total volume, ft^3

   This value includes the cold plate volume, the volume of the piping in the equipment loop, and the total volume (piping and heat exchanger) of the metabolic loop.

   TEMP(IMOD,4) = acquisition surface area, ft^2

   This value includes only the total surface area of the conductive cold plates in the equipment loop.

   TEMP(IMOD,5) = total cold plate load, kW

If the equipment loop is integrated, the bus heat exchanger used to couple the equipment loop to the main transport system is considered to be a part of the main transport system. On the other hand, if the equipment loop is autonomous, the weight, volume, etc. of a bus heat exchanger and a body-mounted radiator are included in the totals for the module's equipment loop. These values, however, are computed as part of the acquisition system analysis.

EQUIPMENT LOOPS WITH TWO-PHASE COLD PLATES (Subroutine CANDA2)

Equipment loops with two-phase cold plates employ a working fluid that changes phase from liquid to vapor as it passes through the cold plates. The analysis of these loops is performed in subroutine CANDA2 as outlined below:

1. The metabolic loop is analyzed using subroutine METLOOP to determine the volume, mass, and pump power for the metabolic loop.
2. The two-phase cold plates in the equipment loop are analyzed using subroutine TPCP to determine the mass flow rates through each cold plate, the mass flow rates through each segment of the liquid supply and vapor return lines, the total acquisition surface area, the total cold plate mass, and the total cold plate volume.
3. The liquid supply lines and the branch supply lines are sized using subroutine LIQLINE to determine the pipe mass, the fluid mass, the piping volume, and the total liquid pressure drop in the equipment loop. (The pressure drop through each cold plate is assumed to be 5 psi.)

4. The vapor return lines and the branch return lines are sized using subroutine VAPLINE to determine the pipe mass, the fluid mass, the piping volume, and the total vapor pressure drop in the equipment loop.
5. The total pump power requirement for the equipment loop is determined in subroutine DELPRS.
6. The weight of the pump package for the equipment loop and for the metabolic loop is computed.
7. The results of these analyses are stored in the TEMP array in the following order with IMOD denoting the module number or index:

   TEMP(IMOD,1) = pump power required, kW

   This value includes the pump power required for the equipment loop and the pump power required by the metabolic loop.

   TEMP(IMOD,2) = total mass, lb

   This value includes the cold plate mass, the dry pipe mass and the fluid mass of the equipment loop, the total mass (wet pipe and heat exchanger) of the metabolic loop, and the pump package weight for the equipment loop and for metabolic loop.

   TEMP(IMOD,3) = total volume, ft^3

   This value includes the cold plate volume, the volume of the piping in the equipment loop, and the total volume (piping and heat exchanger) of the metabolic loop.

   TEMP(IMOD,4) = acquisition surface area, ft^2

This value includes only the total surface area of the two-phase cold plates in the equipment loop.

TEMP(IMOD,5) = total cold plate load, kW

If the equipment loop is integrated, the bus heat exchanger used to couple the equipment loop to the main transport system is considered to be a part of the main transport system. On the other hand, if the equipment loop is autonomous, the weight, volume, etc. of a bus heat exchanger and a body-mounted radiator are included in the totals for the module's equipment loop. These values, however, are computed as part of the acquisition system analysis.

PUMPED LIQUID TRANSPORT SYSTEM (Subroutine CANDT1)

In the pumped liquid transport system the working fluid remains in the liquid phase throughout. Integrated modules are coupled to the transport system by bus heat exchangers, and a separate bus heat exchanger couples the main transport loop the main radiator system. The analysis of this loop is performed in subroutine CANDT1 as outlined below:

1. The operating temperature of the transport loop is assumed to be 5°C less than the minimum working fluid temperature in any of the integrated modules.
2. The total heat load of each of the integrated modules determines the load that must be handled by each of the bus heat exchangers. With these loads as well as the working fluids used in each of the integrated modules known, subroutine BUSHX is used to analyze each bus heat exchanger to determine its volume and mass.

3. The total load carried by the transport system is the sum of the integrated module equipment loads. With this load and the radiator working fluid known, subroutine BUSHX is used to analyze the radiator bus heat exchanger to determine its volume and mass.
4. The liquid supply lines, the liquid return lines, and the branch lines to the modules are sized using subroutine LIQLINE to determine the pipe mass, the fluid mass, the piping volume, and the liquid pressure drop in the transport loop. (The pressure drop through each bus heat exchanger is assumed to be 5 psi.)
5. The total pump power requirement for the transport loop is determined in subroutine DELPRS.
6. The weight of the pump package for the transport loop is computed.
7. The results of these analyses are stored in the TEMP array in the following order with the first index of the array denoting the transport system:

   TEMP(8,1) = pump power required, kW

   TEMP(8,2) = total mass, lb

   This value includes the mass of all bus heat exchangers, the dry pipe mass and the fluid mass of the transport loop, and the pump package weight for the transport loop.

   TEMP(8,3) = total volume, ft^3

   This value includes the volume of all bus heat exchangers and the volume of the piping in the transport loop.

   TEMP(8,5) = total transport system load, kW

TWO-PHASE TRANSPORT SYSTEM (Subroutine CANDT2)

In the two-phase transport system the working fluid changes phase as it passes through the bus heat exchangers. Integrated modules are coupled to the transport system by bus heat exchangers; and a separate bus heat exchanger couples the main transport loop the main radiator system. The analysis of this loop is performed in subroutine CANDT2 as outlined below:

1. The operating temperature of the transport loop is assumed to be 5°C less than the minimum working fluid temperature in any of the integrated modules.
2. The total heat load of each of the integrated modules determines the load that must be handled by each of the bus heat exchangers. With these loads as well as the working fluids used in each of the integrated modules known, subroutine BUSHX is used to analyze each bus heat exchanger to determine the volume and mass of each.
3. The total load carried by the transport system is the sum of each of the integrated module equipment loads. With this load and the radiator working fluid known, subroutine BUSHX is used to analyze the radiator bus heat exchanger to determine its volume and mass.
4. The liquid supply lines and the liquid branch lines to the modules are sized using subroutine LIQLINE to determine the pipe mass, the fluid mass, the piping volume, and the liquid pressure drop in the transport loop. (The pressure drop through each bus heat exchanger is assumed to be 5 psi.)

5. The vapor return lines and the vapor branch lines from the modules are sized using subroutine VAPLINE to determine the pipe mass, the fluid mass, the piping volume, and the vapor pressure drop in the transport loop.
6. The total pump power requirement for the transport loop is determined in subroutine DELPRS.
7. The weight of the pump package for the transport loop is computed.
8. The results of these analyses are stored in the TEMP array in the following order with the first index of the array denoting the transport system:

   TEMP(8,1) = pump power required, kW

   TEMP(8,2) = total mass, lb

   This value includes the mass of all bus heat exchangers, the dry pipe mass and the fluid mass of the transport loop, and the pump package weight for the transport loop.

   TEMP(8,3) = total volume, ft^3

   This value includes the volume of all bus heat exchangers and the volume of the piping in the transport loop.

   TEMP(8,5) = total transport system load, kW

METABOLIC LOOP (Subroutine METLOOP)

The metabolic loop is assumed to be composed of a single, pumped liquid water loop operating at 25°C. An air/water heat exchanger is used to cool the cabin air, and the heat is rejected at each module by a body-mounted radiator.

The mass flow rate of water is determined from the metabolic load assuming that the water experiences a 20°C increase in temperature as it passes through the heat exchanger. The volume of the air/water heat exchanger is sized by assuming that 1 ft^3 is required for each 2.36 kW of metabolic load, and the mass of the heat exchanger is assumed to be 4.92 lb/kW.

The liquid line for the metabolic loop is sized using subroutine LIQLINE, which also computes the wet and dry line weights and the fluid pressure drop. The pump power required is computed in subroutine DELPRS.

The volume and weight of the bus heat exchanger, which couples the metabolic loop to the body-mounted radiator, are determined in subroutine BUSHX. The volume and weight of the radiator are computed in subroutine CANDR1 (heat pipe radiator analysis).

The mass computed in METLOOP consists of the air/water heat exchanger mass, the bus heat exchanger mass, and the wet mass of the pipe. The volume is determined from the sum of the volumes of each of these components.

SUMMARY

The orbiting space station being developed by the National Aeronautics and Space Administration will have many thermal sources and sinks as well as requirements for the transport of thermal energy through large distances. The station is also expected to evolve over twenty or more years from an initial design. As the station evolves, thermal management will become more difficult. Thus, analysis techniques to evaluate the effects of changing various thermal loads and the methods utilized to control temperature distributions in the station are essential.

Analysis techniques, including a user-friendly computer program, have been developed which should prove quite useful to thermal designers and systems analysts working on the space station. The program uses a data base and user input to compute costs, sizes and power requirements for individual components and complete systems. User input consists of selecting mission parameters, selecting thermal acquisition configurations, transport systems and distances, and thermal rejection configurations. The capabilities of the program may be expanded by including additional thermal models as subroutines.

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APPENDICES

APPENDIX A
DATA BASE CONTENTS

Record No.	Format	Variable Names
1	(215,11A10)	NOSYS,NOREC,(NAMES(I),I=1,11)
2-6	(12A10)	(NAMES(I),I=12*J,12*J+11) J ranges from 1 to 5 as record number changes
7	(15F8.3)	(RMISION(I),I=1,15)
8-22	(12F10.6)	(CANDAT(IMOD,I),I=1,12) IMOD ranges from 1 to 15 as record number changes

System configuration file 1 ;(i.e. NAMES(1) - default configuration)

23	(A10,A6,A34,A70)	NAME,DATE,PREPARE,TITLE
24-30	(20F6.2)	(MODDATA(N,J),J=1,20) N ranges from 1 to 7 as record number changes
31	(15F8.2)	(MODDATA(8,J),J=1,15)
32-38	(7A4,14F6.2,4A2)	(SYSNAM(N,J),J=1,7) (SYSDATA(N,J),J=1,8), (SYSDATA(N,J),J=1,15), PMATL(N),PMATL(N+7),PMATL(15), PMATL(16) N ranges from 1 to 7 as record number changes
39	(7A9,A53)	(MODULE(J),J=1,7),DUMNAME

System configuration file 2; (i.e. NAMES(2))

17 records for each configuration, arranged as described above for the default configuration. Each subsequent block of 17 records contains a separate system configuration file.

VARIABLE DEFINITIONS

NOSYS	number of system configuration files in the data base
NOREC	number of records required for each system configuration file
NAMES(I)	name of system configuration file I
RMISION(I)	mission model parameter file
I=1	not used
I=2	mission duration, days
I=3	resupply interval, days
I=4	power penalty, lb/kW
I=5	control penalty, lb/kW
I=6	propulsion penalty, lb/kW
I=7-10	not used
I=11	probability of meteroid penetration
I=12	transportation cost factor, k$/lb
I=13	maintenance cost factor, k$/lb
I=14	integration cost factor, %
I=15	programmatic cost factor, %
CANDDAT(IMOD,I)	candidate data file for candidate having index IMOD (IMOD=1-5 for five acquisition candidates, IMOD=6-10 for five transport candidates, IMOD=11-15 for five rejection candidates)
I=1	weight of spares for 90 days, lb
I=2	volume of spares for 90 days, ft^3
I=3	weight of consumables for 90 days, lb
I=4	volume of consumables for 90 days, ft^3
I=5	reliability (0-8)
I=6	technology readiness (0-8)
I=7	pacing technology problems (0-8)
I=8	90 day maintenance time, hr
I=9	nonrecurring design, development, test and certify, 1983 million $
I=10	spares and consumables to operate for 90 days, 1983 million $
I=11	cost of flight unit, 1983 million $
I=12	candidate rating, kW
MODDATA(IMOD,I)	cold plate location data for module IMOD (<8)
I=1-5	supply line lengths (ft) for CP 1-5
I=6-10	branch supply lengths (ft) for CP 1-5
I=11-15	return line lengths (ft) for CP 1-5
I=16-20	branch return lengths (ft) for CP 1-5

MODDATA(8,I)	transport lengths to modules
I=1,3,4,7,9,11,13	length (ft) from main radiator to modules 1-7
I=2,3,6,8,10,12,14	branch length (ft) to modules 1-7
SYSNAME(IMOD,I)	
I=1	either "AUTO" for autonomous or "INTG" for integrated
I=2	either "CCP" or "TPCP" or "CPCP" - cold plate candidate abbreviations
I=3	either "PLL" or "PTPL" or "HHPR" - transport candidate abbreviations
I=4	either "HPR or "HHPR" or "LDR" - rejection candidate abbreviations
I=5	either "WATE" or "AMMO" or "F-11" - equipment loop working fluid abbreviations
I=6	either "WATE" or "AMMO" or "F-11" - transport loop working fluid abbreviations
I=7	either "WATE" or "AMMO" or "F-11" or "ACET" or "METH" - rejection system working fluid abbreviations
SYSDATA(IMOD,I)	system configuration data for module IMOD
I=1	number of active cold plates (<6)
I=2	cold plate operating temperature, ^{o}C
I=3	metabolic load, kW
I=4-8	loads, kW, for cold plates 1-5
I=9-11	not used
I=12	radiator surface temperature, ^{o}C
I=13	emissivity of radiator surface
I=14	absorptivity of radiator surface
I=15	heat pipe radiator operating temperature, ^{o}C
PMATL(I)	material types - either "AL" or "SS"
I=1-7	material type for cold plates and pipe in modules 1-7
I=8-15	material type for radiators of modules 1-7
I=16	material type for transport loop
MODULE(I)	names for modules 1-7 (max 9 characters)

APPENDIX B

ASSESSMENT ALGORITHMS

Acquisition Assessment Algorithms for Individual Modules

A. Reliability, Technology Readiness and Pacing Technology Rating:

For integrated Modules

$$\begin{Bmatrix} R_i \\ TR_i \\ PT_i \end{Bmatrix} = \begin{Bmatrix} R_{c,a} \\ TR_{c,a} \\ PT_{c,a} \end{Bmatrix}$$

For autonomous modules

$$\begin{Bmatrix} R_i \\ TR_i \\ PT_i \end{Bmatrix} = \begin{Bmatrix} \text{Minimum } (R_{c,a},\ R_{c,t},\ R_{c,r}) \\ \text{Minimum } (TR_{c,s},\ TR_{c,t},\ TR_{c,r}) \\ \text{Minimum } (RT_{c,a},\ PT_{c,t},\ PT_{c,r}) \end{Bmatrix}$$

B. Metabolic Load

$ML_i = ML_i$ from system configuration file, $i = 1,\ldots,n$

C. Acquisition Load

$$AL_i = \sum_{j=1}^{p} (CP_j)_i \; ; \; i = 1,\ldots,n$$

ML_T = sum of AL_i for integrated modules

$ML_R = ML_T$

D. Resupply consumables

$$RC_i = RC_m + (WS_a + WC_a) \left(\frac{AL_i}{CR_a} \right) \left(\frac{RI}{90} \right) \text{ for integrated modules}$$

$$RC_i = RC_m + \left[\sum_{k=e,t,r} (WS_k + WC_k)/CR_k \right] (AL_i) \left(\frac{RI}{90} \right) \text{ for autonomous modules}$$

$$RC_k = (WS_k + WC_k) \left(\frac{ML_k}{CR_k} \right) \left(\frac{RI}{90} \right) \; ; \; k = T,R$$

E. Resupply Volume

$$RV_i = RV_m + (VS_a + VC_a) \left(\frac{AL_i}{CR_a} \right) \left(\frac{RI}{90} \right) \text{ for integrated modules}$$

$$RV_i = RV_m + \left[\sum_{k=a,t,r} (VS_k + VC_k)/CR_k \right] (AL_i) \left(\frac{RI}{90} \right) \text{ for autonomous modules}$$

$$RV_k = (VS_k + VC_k)\left(\frac{ML_k}{CR_k}\right)\left(\frac{RI}{90}\right)$$

F. Power Required

PR_i = external power requirement of TCS for module (or main transport/main rejection system) computed in candidate subroutine; i = 1,...,n and T,R (note 1)

G. Power System Impact

$PSI_i = (PR_i)(PSP)$; i = 1,...,n and T,R

H. Control System Impact

$CSI_i = (PR_i)(CSP)$; i = 1,...,n and T,R

I. Propulsion System Impact

$PRSI_i = (PR_i)(PRSP)$; i = 1,...,n and T,R

J. Launch Weight

LW_i = launch weight of TCS for module (or main transport/rejection system) computed in candidate subroutine; i = 1,...,n and T,R (Note 1)

K. Launch Volume

LV_i = launch volume of TCS for module (or main transport, rejection system) computed in candidate subroutine; i = 1,...,n and T,R (Note 1)

L. Equivalent Launch Weight

$ELW_i = RC_i + PSI_i + CSI_i + PRSI_i + LW_i$; i = 1,...,n and T,R

M. Maintenance Time Over Resupply Interval

$$MT_i = MT_m + (RMT_a)\left(\frac{AL_i}{CR_a}\right)\left(\frac{RI}{90}\right) \text{ for integrated modules}$$

$$MT_i = MT_m + \left[\sum_{k=a,t,r} (RMT_k)/CR_k\right](AL_i)\left(\frac{RI}{90}\right) \text{ for autonomous modules}$$

$$MT_k = (RMT_k)\left(\frac{MT_k}{CR_k}\right)\left(\frac{RI}{90}\right); \qquad k = T,R$$

N. Acquisition Surface Area

ASA_i = total cold plate surface area for modules computed in candidate subroutine; $i = 1,\ldots,n$.

O. Rejection Surface Area

RSA_i = RSA_m + rejection surface area for autonomous module (or main rejection system) computed in candidate subroutine; i = autonomous modules and R.

Note: The following costs are FY83 million dollars.

P. Cost of Design, Development, Test and Evaluate

$CDTE_i$ = $(DDTE_a)$/(number of modules having same acquisition candidate) $i = 1,\ldots,n$

$CDTE_k$ = $(DDTE_k)$/(number of modules having same k candidate + 1); $k=T,R$

Q. Cost of Flight Unit, Spares and Consumables for Initial Launch

$$CFU_i = \left[FU_a + (CSC_a)\left(\frac{RI}{90}\right)\right]\left(\frac{AL_i}{CR_a}\right); \qquad i = 1,\ldots,n \text{ (Note 1)}$$

$$CRU_R = \left[FU_k + (CSC_k)\left(\frac{RI}{90}\right)\right]\left(\frac{ML_k}{CR_k}\right); \qquad k = T,R$$

R. Cost of spares and consumables to operate over mission

$$CSC_i = (CS_a)\left[\frac{MD}{RI} - 1\right]\left(\frac{AL_i}{CR_a}\right); \qquad i = 1,\ldots,n \quad \text{(Note 1)}$$

$$CSC_k = (CS_k)\left(\frac{MD}{RI} - 1\right)\left(\frac{ML_k}{CR_k}\right); \qquad k = T,R$$

S. Integration Cost

$$CI_i = (CDTE_i + CFU_i)(ICF/100); \quad i = 1,...,n \text{ and } T,R$$

T. Programmatic Cost

$$CPR_i = (CDTE_i + CFU_i)(PCF/100); \quad i = 1,...,n \text{ and } T,R$$

U. Transportation Costs for a Spares and Consumables Over Mission

$$CTSC_i = (RC_i)\left[\frac{MP}{RI} - 1\right](TCF/1000); \quad i = 1,...,n \text{ and } T,R$$

V. Transportation cost for flight unit, spares and consumables to operate over initial resupply interval

$$CTFU_i = (RC_i + LW_i)(TCF/1000); \; i = 1,...,n \text{ and } T,R$$

W. Cost of Maintenance for Mission

$$CMM_i = (MT_i)\left[\frac{MD}{RI} - 1\right]\left[\frac{MCF}{1000}\right]; \quad i = 1,...,n \text{ and } T,R$$

X. Life Cycle Cost for Mission

$$CLC_i = (CDTE_i + CFU_i + CCS_i + CI_i + CPR_i + CTSC_i + CTFU_i + CMM_i) \quad ;$$

$$i = 1,...,n \text{ and } T,R$$

Note 1: Includes only acquisition system for integrated modules; includes acquisition, transport and reject systems for autonomous modules.

II. Summary Assessment Algorithms

A.
$$\begin{Bmatrix} R_A \\ TR_A \\ PT_A \end{Bmatrix} = \begin{Bmatrix} \text{Minimum } (R_i;\ i = 1,...,n) \\ \text{Minimum } (TR_i;\ i = 1,...,n) \\ \text{Minimum } (PT_i;\ i = 1,...,n) \end{Bmatrix}$$

$$\begin{Bmatrix} R_o \\ TR_o \\ PT_o \end{Bmatrix} = \begin{Bmatrix} \text{Minimum } (R_k;\ k = A,\ T,\ R) \\ \text{Minimum } ({}_R k;\ k = A,\ T,\ R) \\ \text{Minimum } (R_k;\ k = A,\ T,\ R) \end{Bmatrix}$$

B. $ML_A = \sum_{i=1}^{n} ML_i$; and $ML_o = ML_A$

C. AAL = Sum of AL_i for autonomous modules

IAL = Sum of AL_i for integrated modules

D. through X.

$$Value_A = \sum_{i=1}^{n} Value_i$$

$$Value_o = Value_A + Value_T + Value_R$$

NOMENCLATURE FOR APPENDIX C

AAL	autonomous acquisition load, kW
ACDF	acquisition candidate data file
AL	acquisition load, kW
ASA	acquisition surface area, ft^2
CDTE	cost of design, development, test and evaluation, million $
CFU	cost of flight unit, spares, and consumables for initial launch, million $
CI	integration cost, million $
CLC	life cycle cost for mission, million $
CP	cold plate load, kW
CR	candidate rating, kW, from ACDF
CS	cost of spares and consumables for 90 days from ACDF, million $
CSC	cost of spares and consumables to operate over mission, million $
CSI	control system impact, lb
CSP	control system penalty, lb/kW, from MMPF
CTFU	transportation cost for flight unit, spares and consumables to operate over initial resupply interval, million $
CTSC	transportation cost for spares and consumables over mission, million $
DDTE	design, development, test and evaluate cost from ACDF, million $
FU	flight unit cost for candidate from ACDF, million $
IAL	integrated acquisition load, kW
ICF	integration cost factor, %, from MMPF
LV	launch volume, ft^3
LW	launch weight, lb

MCF	maintenance cost factor, k$/hr, from MMPF
MD	mission duration, days, from MMPF
ML	metabolic load, kW
MMPF	mission model parameter file
MT	maintenance time over resupply interval, hr
PCF	programmatic cost factor, %, from MMPF
PR	power required, kW
PRSI	propulsion system impact, lb
PRSP	propulsion system penalty, lb/kW, from MMPF
PSI	power system impact, lb
PSP	power system penalty, lb/kW, from MMPF
PT	pacing technology rating
R	reliability
RC	resupply consumables, lb
RI	resupply interval, days, from MMPF
RMT	90-day maintenance time, hr, from ACDF
RSA	rejection surface area, ft^2
RV	resupply volume, ft^3
TCF	transportation cost factor, k$/lb from MMPF
TR	technology readiness
VC	volume of consumables from 90 days, ft^3, ACDF
VS	volume of spares for 90 days, ft^3, ACDF
WC	weight of consumables for 90 days, lb, from ACDF
WX	weight of spares for 90 days, lb, from ACDF

Subscripts

a	acquisition candidate
A	total acquisition system
c	candidate data file value
i	module i
j	cold plate
m	metabolic loop
n	number of modules
o	overall assessment
p	number of cold plates
r	rejection candidate
R	main rejection system
t	transport candidate
T	main transport system